

Development of Novel Methods for the Reduction of Noise and Weight in Helicopter Transmissions.

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Introduction

Over the 70-year evolution of the helicopter, man's understanding of vibration control has greatly increased. However, in spite of the increased performance, the extent of helicopter vibration problems has not significantly diminished. Crew vibration and noise remains important factors in the design of all current helicopters. With more complex and critical demands being placed on aircrews, it is essential that vibration and noise not impair their performance.

A major source of helicopter cabin noise (which has been measured at a sound pressure level of over 100 dB) is the gearbox. Reduction of this noise has been a goal of NASA and the U.S. Army. Gear mesh noise is typically in the frequency range of 1000 to 3000 Hz, a range important for speech. A requirement for U.S. Army/NASA Advanced Rotorcraft Transmission project has been a 10-dB reduction compared to current designs. A combined analytical/experimental effort has been underway, since the end of the 80's, to study effects of design parameters on noise production, Refs. 1 to 6.

The noise generated by the gear mesh can be transmitted to the surrounding media through the bearings that support the gear shaft (Ref. 2). Therefore, the use of fluid film bearings instead of rolling element bearings could reduce the transmission noise by 10 dB, Refs. 7 and 8. In addition, the fluid film bearings that support the gear shaft can change the dynamics of the gear assembly by providing damping to the system and by being softer than rolling element bearings. A novel fluid film bearing - the wave bearing, invented by Dr. Florin Dimofte, Refs. 9 and 10, is an excellent candidate to replace rigid rolling element bearings and damp the noise/vibrations that cross the bearing to the housing. Moreover, by use of the wave bearing the dynamic stiffness and damping can be adjusted over a rather large range (estimated to be from one to two orders of magnitude) to match the dynamic needs of the gear system. This will diminish the noise/vibrations at the source - where the gear mesh is engaged.

A new method to evaluate the rate of noise that passes through the wave bearing has been recently developed. This method is based on measurements taken, using a stethoscope, at various locations on a bearing tested on the Wave Bearing Rig built at the NASA Glenn Research Center (GRC) in Cleveland, Ohio with funds from the General Aviation Program (GAP). This method can provide a

better understanding of the physical phenomena involved. Subsequently, accelerometers are mounted at the same locations and wave signals and FFT analysis is used. This is further described in Section 1: "Noise Measurements on the Wave Bearing Rig."

In Refs. 4 - 6, and also in other reports, a NASA GRC Gear Noise Rig has been used. This rig has a generic gearbox with two spur gears mounted on the input and output shafts, respectively. These shafts are supported by ball bearings. To quantify the noise attenuation by the wave bearings and to compare with the noise generated by this gearbox when ball bearings are used, the gear box has been re-designed and the ball bearings were replaced by journal and thrust wave bearings. This is further described in Section 2: "Re-design of the Generic Gearbox with Wave Bearings."

To assess the use of wave bearings in a helicopter transmission, preliminary computation of the bearing performance has been performed. Seven potential bearings were selected and the results are presented in Section 3: "Preliminary Computation to Assess the Use of the Wave Bearings in Helicopter Transmissions."

The noise/vibrations generated by the two engaged gear meshes could be attenuated if the engaged mesh surfaces have a high quality finish. A root-mean average (RMA) of 2 micro-inches or less should be just enough to reduce the noise/vibration and to increase the life of the gears. Smaller RMA values are better. A near-mirror surface quality could be produced by several methods such as abrasive polishing, chemical assisted polishing, or grinding. Production of the surfaces by grinding offers the possibility to control the macro-geometry (i.e., the form), waviness, and surface texture with one process. Therefore, an effort to demonstrate a precise grinding process in-house at NASA GRC was organized and conducted. Results are presented in Section 4: "New Grinding Technique to Produce Near-Mirror Surface Quality."

Using a wave bearing in a helicopter transmission will also result in the bearing performing well during starts and stops. The bearing should have a lifting speed as low as possible. At the lifting speed, the fluid film separates the bearing surfaces. But the bearing must also resist wear when the speed is less than the lifting speed and the bearing surfaces are in direct contact. Coatings are used to overcome this problem. Encouraging results have been obtained when the wave bearing was tested under the GAP program. In fact, the wave bearings have been run through more than 700 start-stop cycles and have been run for 15 minutes without oil. But new tests are planned involving new commercially available coatings to match the helicopter industry request of a 30-minute run time without any oil. The preparation for such testing is reported in Section: "Preliminary Work for Coating Tests."

1. Noise Measurements on the Wave Bearing Rig

A new method to evaluate the rate of noise that is passing through the wave bearing has been developed. This method is based on measurements at various locations on the bearing to determine local noise. The approach was applied to a wave bearing on the Wave Bearing Rig built during research for GAP. However, the rig was modified from the original GAP design. Thus, the ball bearings that supported the shaft were replaced by wave bearings and four checkpoints were setup along the turbine-rig assembly:

1. on the turbine housing - named TURBINE,
2. on the rig housing close to the shaft wave bearing near the turbine - named TURBINE SIDE,
3. on the rod that is used to load the test bearing - named TEST BEARING,
4. on the rig housing close to the shaft wave bearing opposite the turbine - named DOOR SIDE.

Figure 1 shows the test rig prepared for measurements and all of the checkpoints are labeled.

a. measurements by using an electronic stethoscope device

An electronic stethoscope was modified to accurately measure the noise level at various locations along the turbine-rig shafts and to listen to the noise through headphones as well (Figure 2). This method to analyze the noise provides a better understanding of the physical phenomena involved. A precise potentiometer was added to control the output level of the stethoscope amplifier (Figure 1, No.1). This potentiometer permitted the stethoscope to be employed at the same setup location each time the data is read and enabled a consistent comparison between data to be made. Next, the measuring port (Figure 2, No.2) was electronically separated from the listening port (Figure 2, No.3) and the volume of the noise can be adjusted with the original volume knob without changing the measurement setup. Figure 2 shows the entire stethoscope assembly with a multimeter and the headphones connected to measuring port, No.2, and the listening port, No.3, respectively.

The noise and its level were recorded at three running speeds: 6,000, 12,000, and 18,000 RPM. At each speed the test bearing ran unloaded and loaded as well. The loads were 100, 200, and 300 lbs, respectively to the speeds. Figure 1 shows the rig when run at 18,000 RPM and loaded with 300 lbs. The stethoscope rod is shown touching the turbine housing at checkpoint No.1 in Figure 1. A videotape and a video CD were recorded that shows the check point, the level of the noise on the multimeter screen and the noise that was heard. This tape was presented to the Mechanical Components Branch and then was used in presentations of the wave bearing technology to industry (Pratt & Whitney, Sikorsky, Rolls Royce, etc).

A summary of the noise level measurements with the stethoscope can be seen in Table 1. The noise generated by the turbine was considered as the reference (F_0), and the attenuation at each checkpoint, in dB, was calculated by the relation:

$$dB = 20 \log_{10} \left(\frac{F}{F_0} \right)$$

Table 1 Noise (Sound) Levels, mV, and Attenuation from the Max Level, dB, at Various Check Points

Speed, RPM	Turbine	Turbine Side BRG	Test BRG	Door Side BRG
6,000	400-550 mV 0 dB	22 mV -26.7 dB	20 mV -27.5 dB	34 mV -22.9 dB
12,000	1400-2300 mV 0 dB	26-38 mV -35.2 dB	45 mV -32.3 dB	45-54 mV -31.4 dB
18,000	1400-2000 mV 0 dB	32-37 mV -33.8 dB	47-59 mV -30.12 dB	50-51 mV -30.5 dB

A 23 to 35 dB attenuation was found. The attenuation is higher as the speed is increased.

b. noise /vibration measurements with accelerometers

Four accelerometers were also used to read the noise at locations similar to those that were used when the readings were performed with the stethoscope. These locations and the channel numbers of each location can be seen in the rig cross-section presented in Figure 3. The actual arrangement of these accelerometers on the turbine and rig housings is shown in Figures 4 and 5.

The accelerometer wave signals were pre-amplified by a four-channel preamplifier (Figure 4) and then displayed on the screen of a Tektronix oscilloscope with four channels. The screen can also be recorded on a floppy disc as a graphic file (TIF). A typical view of the scope screen is shown in Figure 6. The RMS value of each channel was displayed on the ESCORT data acquisition screen as well. The Tektronix oscilloscope can also provide a Fast Fourier Transform (FFT) analysis for each channel. Both the wave signal and the FFT analysis results are displayed on the same screen (see Figures 7 and 8).

The bearing was run with KRYTOX 143AD as the lubricant at 6,000, 12,000, and 18,000 RPM. All runs showed, as was typical, a high level of noise generated by the turbine and a strong attenuation of this noise if it was passed through the wave bearings. In addition, the FFT analysis shows that the turbine noise has a considerable number of strong components at higher frequencies than the frequency synchronous to the running speed. A typical example can be seen in Figure 7, where the data from channel 1 - turbine - is displayed. Then, if noise is collected after passing one of the wave bearings, a general attenuation of the noise can be easily observed by checking the RMS values. But FFT analysis also shows that the frequency components of the noise change and most of the noise components at frequencies higher than the synchronous frequency are eliminated. A typical example is shown in Figure 8 where data from channel 3 - test bearing is displayed. The attenuation ratio when the rig was run with KRYTOX 143 AD was approximately 40 dB.

Conclusion

Wave bearings can attenuate, and filter, the noise generated by a machine component due to the dynamic stiffness and damping coefficients. The attenuation ratio could be as large as 35-40 dB. The noise components at higher frequencies than a synchronous frequency can be almost eliminated.

2. Re-design of the Generic Gearbox with Wave Bearings

To consistently quantify the attenuation ratio of the noise/vibration, when wave (fluid) bearings are used instead of ball bearings in a gearbox, plans were made to modify a generic gearbox. It was proposed that noise/vibration measurements with wave and ball bearing should be performed and compared. In Refs. 4-6, and also in other reports, a NASA GRC Gear Noise Rig has been used. This rig has a generic gearbox with two spur gears mounted on the input and output shafts, respectively. These shafts are supported by ball bearings. The gearbox has been re-designed and the ball bearings were replaced by journal and thrust wave bearings.

The original gearbox design can be seen in Figure 9. Four ball bearings were used to support the input and output shaft. A pair of spur gears are mounted at mid-span of the shafts. The ball bearings were replaced by four wave journal-thrust wave bearings. The journal wave bearings have a diameter of 32 mm and are 20 mm long. The radial clearance, C , is 0.018 mm and the nominal wave amplitude ratio (wave amplitude/radial clearance) is 0.3. The thrust bearing has an ID = 34 mm and an OD = 44 mm. A cross section of the wave journal-thrust bearing assembly is shown in Figure 11. The oil supply system was modified to provide lubricant to the new bearings and to collect the out-flow of oil from the bearings. The design was completed, including the manufacture drawings, as well. The drawing numbers are: 59217M04A00 to A013.

The dynamic force of a wave bearing can be written as:

$$F = K x + B \frac{dx}{dt}$$

A parametric numerical study of the direct dynamic stiffness, K_{xx} , and damping, B_{xx} , at 0.0 and 0.4 wave amplitude of the wave journal bearing used in the generic gearbox design was performed. Two radial clearances were selected for this study: 0.015 and 0.020 mm and two wave amplitude ratios, ε_w , 0.0 and 0.4, were used. The study covered four running speeds (800, 2,000, 4,000, and 6,000 RPM) and two loads (34 and 305 lbs) for each speed. The results are listed in Tables 2 and 3 for radial clearances of 0.015 and 0.020 mm, respectively.

Table 2
Radial clearance $C = 15 \mu\text{m}$ $\varepsilon_w = 0.0$ $\varepsilon_w = 0.4$

Speed RPM	Load Lbs	$K_{xx}/10^6$ N/m	$B_{xx}/10^5$ Ns/m	$K_{xx}/10^6$ N/m	$B_{xx}/10^5$ Ns/m
800	34	17.40	3.20	40.70	4.14
	305	826.00	74.50	1120.00	33.90
2000	34	15.00	2.09	66.90	2.64
	305	527.00	17.60	594.00	11.70
4000	34	19.20	2.06	117.00	2.17
	305	377.00	6.61	431.00	6.11
6000	34	24.20	2.12	168.00	2.00
	305	310.00	4.46	402.00	4.46

Table 3
Radial clearance $C = 20 \mu\text{m}$ $\varepsilon_w = 0.0$ $\varepsilon_w = 0.4$

Speed RPM	Load Lbs	$K_{xx}/10^6$ N/m	$B_{xx}/10^5$ Ns/m	$K_{xx}/10^6$ N/m	$B_{xx}/10^5$ Ns/m
800	34	20.00	2.58	32.70	2.60
	305	830.00	81.90	1300.00	32.30
2000	34	11.30	1.06	35.20	1.48
	305	526.00	18.00	642.00	9.54
4000	34	11.60	0.87	54.80	1.06
	305	373.00	6.24	419.00	4.35
6000	34	13.30	0.86	75.90	0.95
	305	307.00	3.47	338.00	3.05

Significant difference in dynamic coefficients can be seen at the lower speeds. At higher speeds, the loads are too light for this bearing, the shaft is running close to its concentric position inside the bearing sleeve and the changes in the bearing geometry have less influence on the dynamic coefficients than at the lower speeds.

Conclusions:

The redesign of the generic gearbox with wave bearings included is ready. The wave amplitude is adjustable; any wave amplitude ratio can be applied to these bearings. Thus, various tests can be performed when the parts are fabricated. A matrix of the test parameters will be established at the appropriate time to cover all potential possibilities offered by the wave bearings. Previous basic measurements will be performed with the generic gearbox operating with ball bearings as well. These measurements will provide the data against which the wave bearing data can be compared.

3. Preliminary Computation to Assess the Use of the Wave Bearings in Helicopter Transmission

To assess the use of wave bearings in a helicopter transmission, preliminary computation of bearing performance has been performed for seven potential bearings. The selected bearings are from: KIOWA, BLACKHAWK, APACHE, CHINOOK, and COMANCHE helicopters transmissions. Seven bearings were analyzed, based on their speed, load, diameter, and length. The wave bearing diameter and length were established based on the space that is now allocated to the actual rolling element bearing. Three oil inlet temperatures are used in the evaluation of the bearing performance: 80, 100, and 120 degrees C. The minimum fluid film thickness in the bearing, friction loss, and oil flow are listed in the Table 4 for each oil inlet temperature.

Conclusions:

The wave bearings can successfully replace rolling element bearings in five of the analyzed cases: 2, and 4 to 7. The wave bearings can be accommodated in the same room that is now allocated to the rolling element bearing. The potential cases are selected mainly based on the minimum oil film thickness in the bearing. A minimum thickness over 2 microns can be safe for the bearing if the bearing materials and manufacturing tolerances are properly selected. Cases 1 and 3 could be also applied but the bearing performance, especially at the highest oil inlet temperature, could be marginal. There is to be mentioned that this analysis was performed under the actual transmission design. In most of the cases a re-design of the whole transmission is recommended to take advantage of all wave bearing benefits.

Table 4 Examples of Wave Oil Bearings that can be used in Helicopter Transmissions

Oil inlet temperature: 80C							Oil inlet temperature: 100C			Oil inlet temperature: 120C			
Helicopter	BRG speed RPM	BRG load lb	BRG D mm	BRG L mm	Hmin micrs	Friction Loss w	Oil Supply l/min	Hmin micrs	Friction Loss w	Oil Supply l/min	Hmin micrs	Friction Loss w	Oil Supply l/min
1. OH-58D KIOWA	327	4,48	69	34	1.5	12	0.015	1.2	10	0.017	1.0	9	0.020
2. UH-60A BLACKHAWK	192.7	13,5	216	85	4.6	105	0.332	3.9	77	0.412	3.2	65	0.518
3. AH-64A APACHE	349.5	13,3	113	57	2.2	58	0.064	1.9	51	0.076	1.6	44	0.092
4. CH47-D CHINOOK (upper)	272.7	22,8	145	103	2.9	93	212	2.3	81	261	1.8	66	0.325
5. CH47-D CHINOOK (lower)	589.4	10,2	171	46	3.0	205	0.715	2.5	168	0.888	2.1	137	1.118
6. RAH-66 COMANCHE (rotor shaft)	355	11,9	388	36	3.3	608	1.545	2.8	489	1.978	2.4	393	2.538
7. RAH-66 COMANCHE (end stage shaft)	3930	6,5	100	27	3.7	914	2.743	3.0	759	3.456	2.5	628	4.372

4. New Grinding Technique to Produce Near-Mirror Surface Quality

The noise/vibrations generated by the two engaged gears can be attenuated if the meshing surfaces are finished to a high quality. An RMA of 2 micro-inches or less is thought to be just enough to reduce noise/vibration and to increase the life of the gears. Smaller RMA values are even better. A near-mirror surface quality could be produced by several methods such as abrasive polishing, chemical assisted polishing, or grinding. Production of the surfaces by grinding offers the possibility to control the macro-geometry (form), waviness, and surface texture with one process.

In August 2001, a demonstration session within the GRC machine-shop was organized to show that a precise mirror like finished surface can be produced by a precise grinding technique. Therefore, a precise grinding spindle was brought to GRC from Romania. The spindle was set on a universal grinding machine made by KELLENBERGER (Figure 12) and internal grinding tests were performed on rings provided by a ball bearing manufacturer. Similar grinding wheels and cooling fluids as the ball bearing manufacture suggested were used. To accelerate the technology transfer process, a grinding specialist from Romania: Ioan Bancescu, was invited to come to GRC. The visit was supported by the US Army, NASA, University of Toledo and NASTEC. Mr. Bancescu worked for two weeks with a NASA technician: Ralph Fekete, to perform grinding tests (Figure 13). Probes were also ground using the original spindle of the machine as well for comparative purposes. A mirror-like surface finish was obtained. Surfaces ground with the precise spindle have an RMA as low as 0.016 to 0.019 micro-millimeters (0.745 micro-inches) unlike the surfaces ground with the original spindle that have an RMA not lower than 0.041 micro-millimeters (1.614 micro-inches). Examples of ground surfaces can be seen in Figure 14. The internal surfaces of two rings look very close to surfaces "cut" on aluminum or brass with a special diamond tool by an "air bearing" machine.

The results of these tests were put together in a white-paper entitled "Production of Near-Mirror Surface Quality by Precision Grinding", attached to the present report. This paper was sent to industry, university, and government researchers. The described grinding technology attracted interest from a number of individuals. Therefore, a gear technology meeting was organized at NASA GRC on September 19, 2002. More than 30 people participated in this meeting from: GRC and Redstone Army Research Lab, Arrow Gear, AT&T, Bell Helicopter, Boeing, Extrude Hone, Gleason Works, IITRI, John Deere, Ohio State University, Pratt & Whitney, Rem Chemical, Rolls-Royce Gear Systems, Sikorsky Aircraft, Timken Co., and US Army AMCOM. After the presentation of this technology, the meeting participants were given an opportunity to observe the grinding demonstration in the NASA GRC Machine Shop. Meeting participants can be seen Figures 15 and 16. Figure 17 shows one of the demo rings that has been removed from the grinding machine. The mirror-like quality finish of the internal surface of this ring can be easily seen due to the lines on the graph under the ring.

Concluding Remarks:

Mirror-like finished surfaces could be made with various processes including grinding. If the surface is circular most of these processes can be applied with a reasonable rate of success and have been

used in the rolling element bearing industry for many years. However, if the surface have a specific profile such as a gear tooth surface or a cam some of these polishing processes would not work properly. Although the surface can have a mirror-like finish quality, the form may not have the needed precision. Therefore, a CNC precise grinding process can be very successfully used to accomplish both goals: precision and excellent surface quality. Work is in progress with both gear grinding and helicopter companies as well as with the US Army and IITRI (Ali Manesh) to use this grinding technology in the gear manufacture process.

5. Preliminary Work for Coating Tests

Using a wave bearing in a helicopter transmission also implies that the bearing will perform well during starts and stops. The bearing should have a lifting speed as low as possible. At lifting speed, a fluid film separates the bearing surfaces. But the bearing must also resist wear for speeds less than the lifting speed when the bearing surfaces are in direct contact. Coatings are used to overcome this problem. Encouraging results have been obtained when the wave bearing was tested under the GAP program. In those tests the wave bearing was subjected to more than 700 start-stop cycles and run 15 minutes without any oil. The coating was Balinit WC/C from BALZERS. In the oil-out test, the turbine, that drives the rig shaft, stopped due to increased friction torque. Nevertheless, the bearing surfaces were found to be in acceptable condition and the bearing was able to run again. Therefore, there is a good chance with such coatings that the goal of the helicopter industry for a "30 minute run without oil" could be realized.

In addition, there are some more coatings, such as Casidiam made by ANATECH LTD, coatings made by United Technologies Research Lab, and diamond films coatings developed at Argonne National Lab, that should be tested as well. To consistently test these coatings under wave bearing running conditions, the test rig should be driven by a new turbine with increased power. Test of pairs of bearing sleeves and rotors with the same coatings should be performed to assess the coating capability to overcome any potential start-stop and/or oil-out problems.

Sleeves and rotors have been ordered and fabricated from PYROWEAR-53 carburized steel (a CARPENTER speciality alloy). Such a sleeve and rotor can be seen in Figures 18 and 19, respectively. A coated sleeve with Balinit coating can be seen in Figure 20. At this point 40 pairs of sleeves and rotors are ready to be coated and the refurbishing process of the Wave Bearing Rig is in progress under a UEET program.

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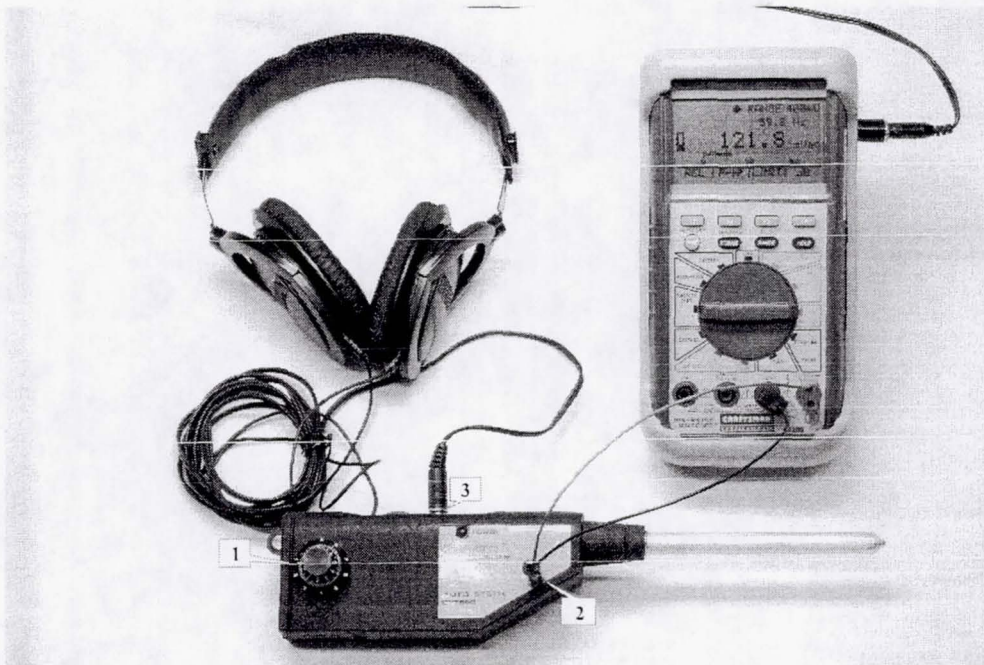


Fig. 1 Modified stethoscope to listen the noise and measure the noise level. A precise potentiometer (1) was added to control the measuring port (2) that was separated from the listening port (3).

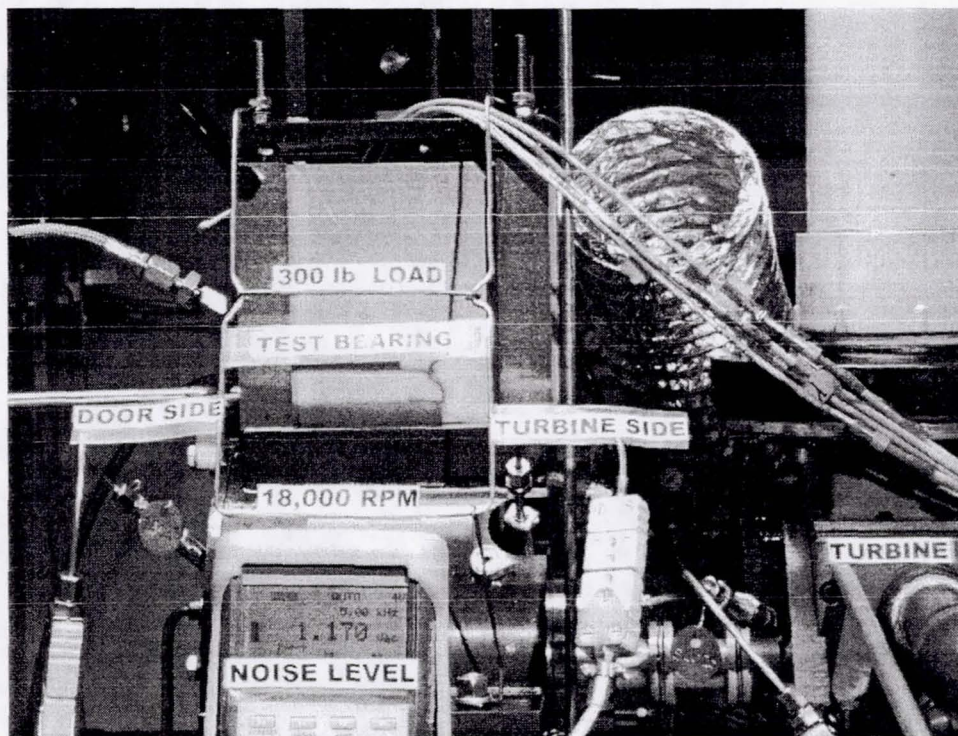


Fig. 2 The checking points with the stethoscope: turbine (housing) , turbine side (wave bearing housing), test bearing (loading rod end), and door side (wave bearing housing). In the picture the stethoscope rod touched the turbine housing.

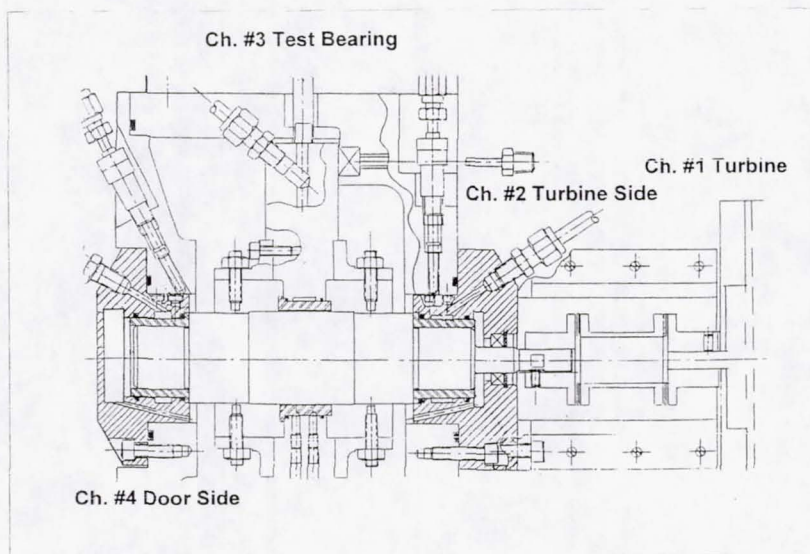


Fig. 3 Accelerometer locations on the rig cross section.

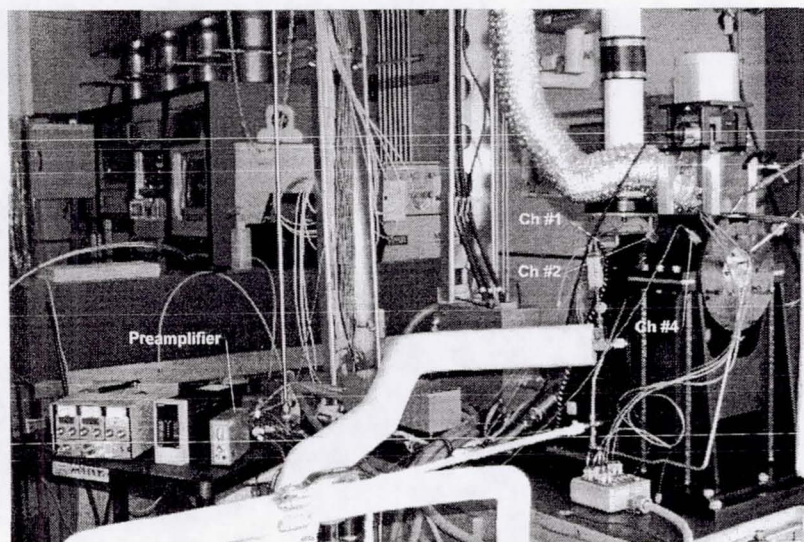


Fig. 4 Positions on the rig of the accelerometers 1, 2, and 4.

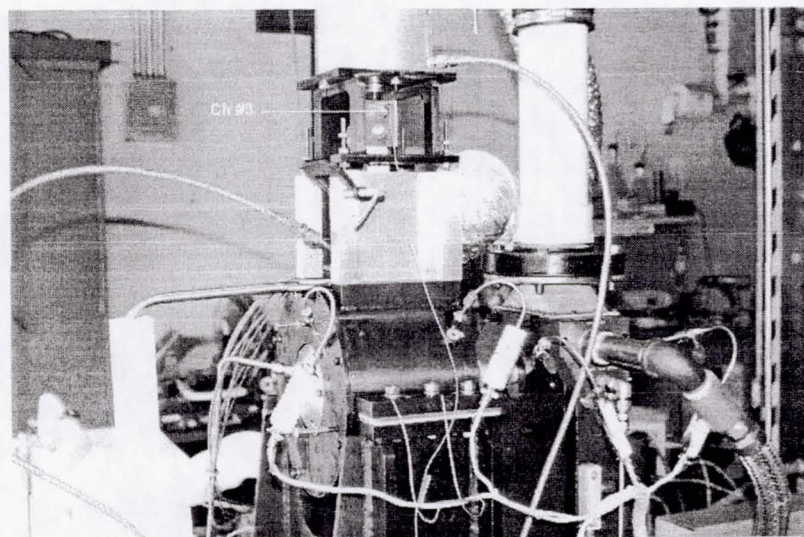


Fig. 5 Position on the rig of the accelerometer 3

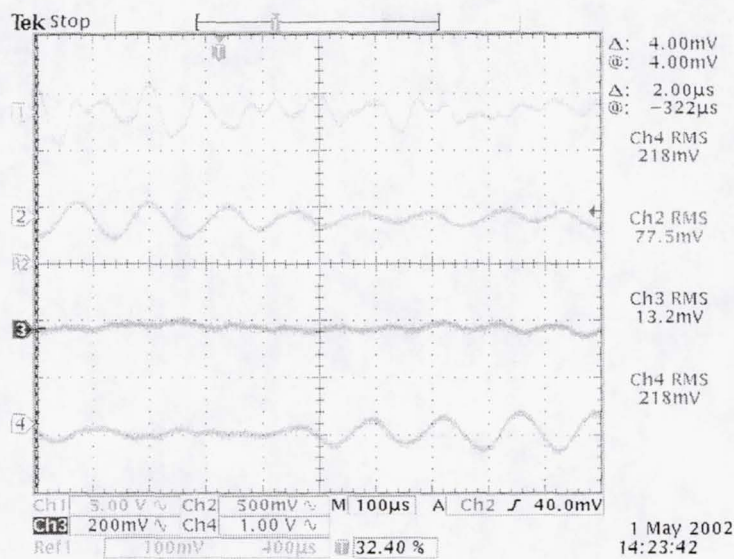


Fig. 6 Wave signals of all four accelerometers at 6,000 RPM rotation speed.

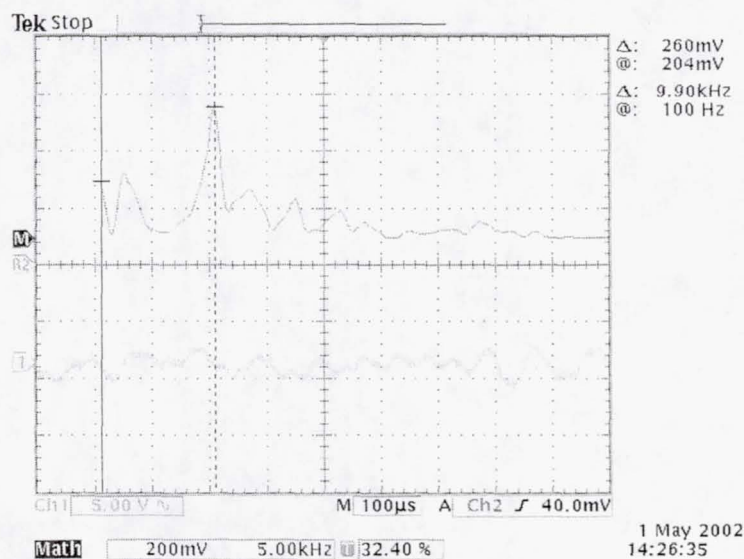


Fig. 7 Wave signal (bottom) and FFT analysis (top) at 6,000 RPM rotation speed for the accelerometer #1 located at the turbine housing.

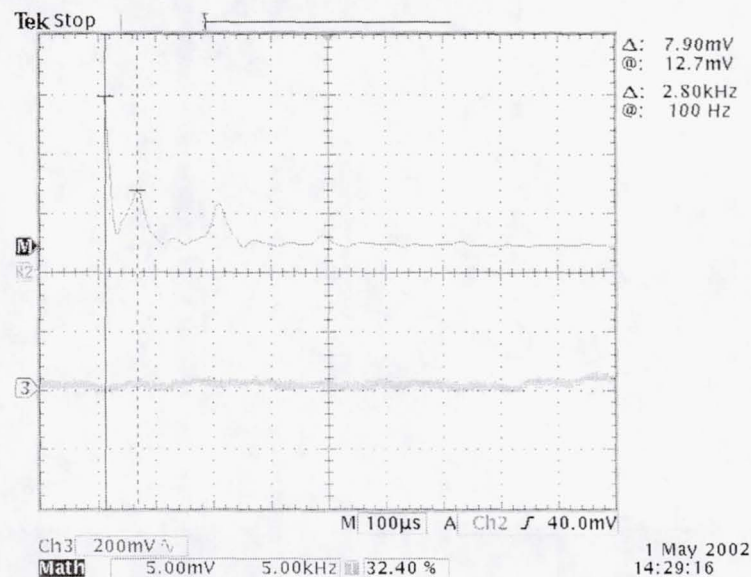


Fig. 8 Wave signal (bottom) and FFT analysis (top) at 6,000 RPM rotation speed for the accelerometer #3 located at the loading rod of the test bearing.

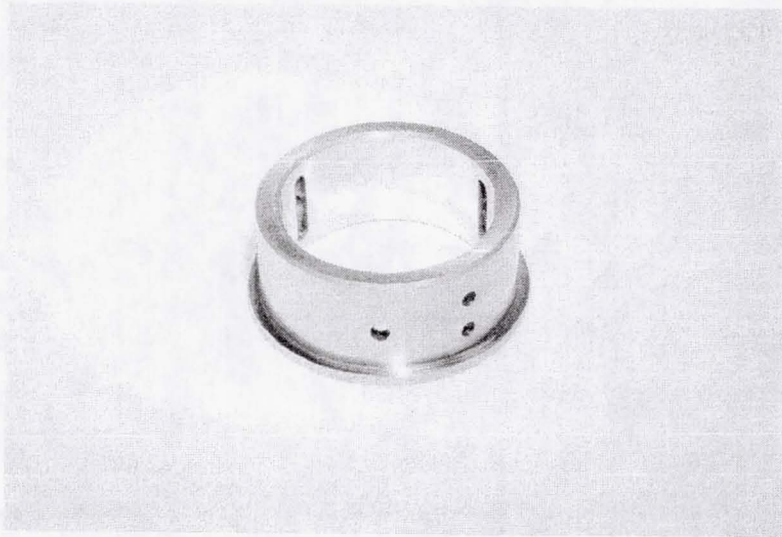


Fig. 3 Wave bearing sleeve from that will be coated and tested for start-stop and oil out running condition.

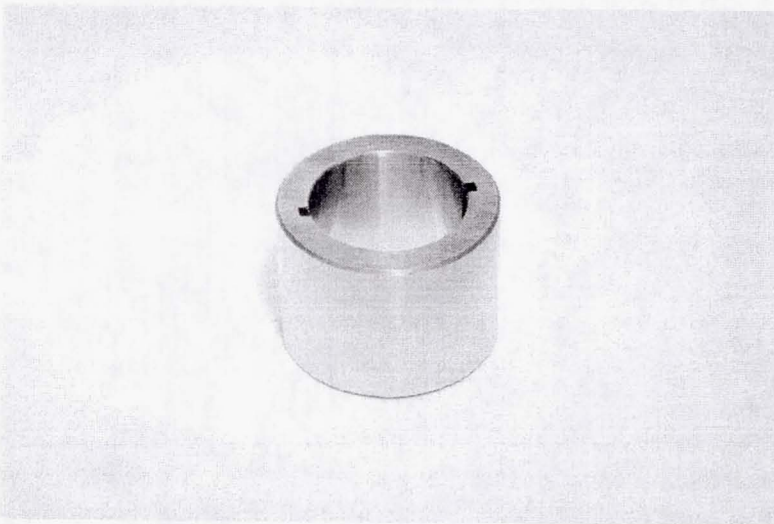
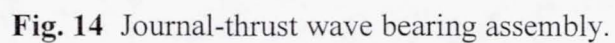
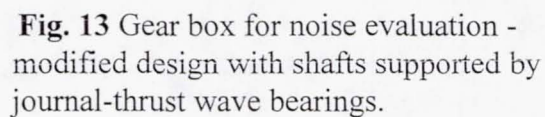
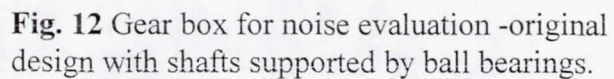


Fig. 4 Wave bearing rotor from that will be coated and tested for start-stop and oil out running condition.



Fig. 5 Example of a coated wave bearing sleeve. The coating is Tungsten-Carbide / Carbon (WC/C)



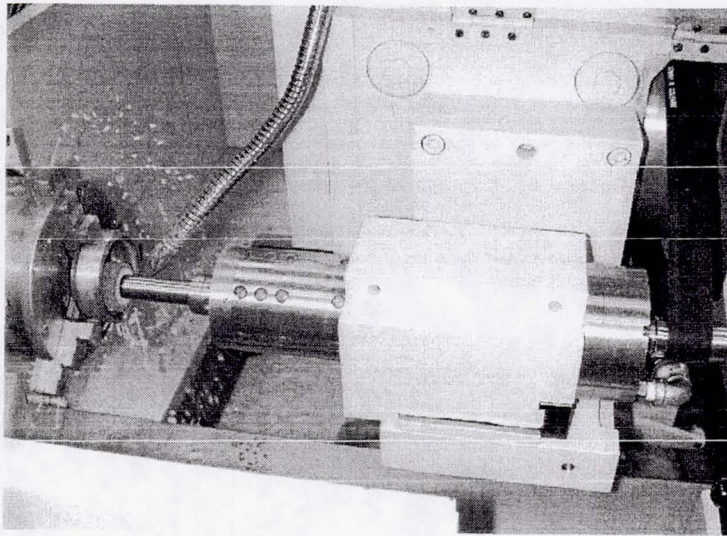


Fig. 15 Precise grinding spindle mounted on a universal grinding machine to demonstrate a mirror like finished grinded surface.

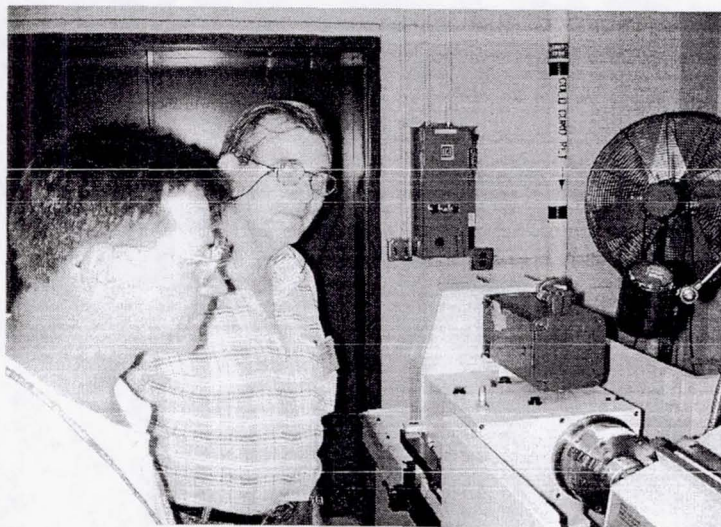


Fig. 16 Grinding technique for mirror like finished surface was transferred to NASA technician by a Romanian grinding specialist in August 2001.

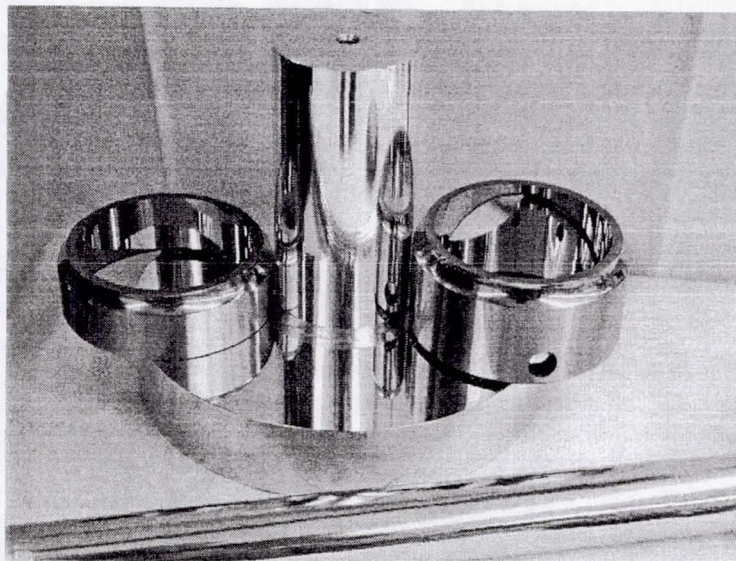


Fig. 17 Comparison between two grinded rings with the inside surface finished like mirror to other parts with similar surface finish quality.



Fig.18 First half of the people who participate to the demo meeting for mirror like surface finish quality by grinding observing the grinding process.



Fig.19 Second half of the people who participate to the demo meeting for mirror like surface finish quality by grinding observing the grinding process.

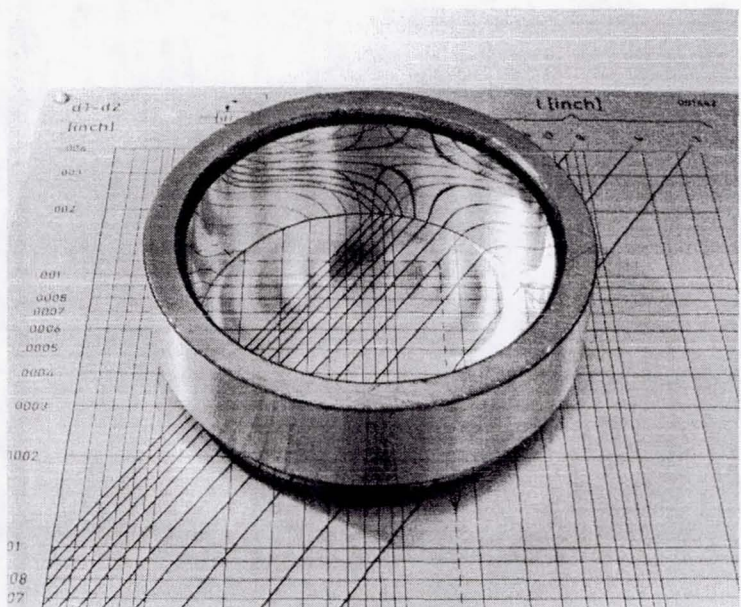


Fig.20 One of the ring shown to the participants to the demo meeting which inside surface was grinded to a mirror like finish quality in front of them.